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PRACTICAL APPLICATION OF A COMPACT, HIGH-EFFECTIVENESS, GAS-TO-GAS, COMPOUND RECUPERATOR WITH LIQUID INTERMEDIARY (CRLI)

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ABSTRACT

Gas-to-gas recuperation with both high thermal effectiveness and order-of-magnitude improvement in cost effectiveness is critical to addressing global energy needs, especially via fuel synthesis from waste CO_2 and renewable H_2. An uncommon compound recuperator with liquid intermediary (**CRLI**) has been simulated for high-ε heat exchange between a first shell-side gas stream and a second shell-side gas stream of similar thermal capacity rates (W/K). The compound recuperator uses a first Gas-to-Liquid (**GL**) recuperator for a nearly complete transfer of available energy from a shell-side gas to an intermediary tube-side heat transfer liquid (**HTL**), followed by a second GL recuperator to transfer the heat from the liquid intermediary to the second gas stream. Each GL recuperator resembles an arrangement of thermally isolated, serially connected, adjacent, cross-flow, finned-tube cores, such as used in AC condensers. They are arranged so to effectively achieve counterflow exchange between the HTL and the shell-side gas stream. The HTL may be water, an organic liquid, a molten alloy, or a molten salt.

Minimization of exergy destruction for the case where there is a substantial temperature difference between the hot and cold sources requires (1) a fairly large number of series connected, thermally isolated cores, (2) similar thermal capacity rates in all three streams (the two gases and the liquid intermediary), (3) a relatively large value for the number of transfer units (**NTU**), and (4) no phase change.

The simulations show that the optimized CRLI recuperator can achieve effectiveness above 97% at very low pumping losses and has the potential for order-of-magnitude reduction in manufacturing costs compared to current technologies for clean gases at pressures above 0.3 MPa at heat transfer rates above ~200 kW.

1. INTRODUCTION

Over the last few decades, many heat exchangers have been well optimized for numerous purposes. However, most have not been directed at very high thermal effectiveness ε for cases where the heat capacity rates in the two gaseous streams are similar. The heat capacity rate *W* of a stream is given by GC_P (its SI units are W/K), where *G* is the mass flow rate (kg/s) and C_P is the specific heat (J/kg-K). By the standard definition of ε (the ratio of heat transferred to the theoretical limit), high ε is most easily achieved when W_{min} (that of the weaker stream) is much less than W_{max} (that of the stronger stream) [*1*]. However, exergy destruction can be minimized only if W_{min} is close to W_{max} [*2*]. The terms "recuperator" and "regenerator" have usually implied the streams have similar *W*'s, and that will be the usage and regime of focus here.

For the high-ε case with similar *W*'s,

$$\varepsilon = 1 - T_\delta /(T_{\mathrm{H}}-T_{\mathrm{C}}) \qquad (1)$$

where T_δ is the mean temperature difference between the streams, T_{H} is the temperature of the hot gas inlet and T_{C} is the temperature of the cold gas inlet.

Common examples of cost-effective heat exchangers with high exergy loss include automobile radiators and air-conditioning condensers. In the automobile radiator, for example, the warmed air leaves at a temperature much below that at which the hot water enters. Thus, most of the water's exergy (energy availability) has been destroyed, irrespective of precisely how one chooses to define it.

The subset of fluid heat exchangers directed at high ε have mostly addressed one of the following cases: condensing-vapor to liquid, condensing-vapor to gas, boiling-liquid to liquid, boiling-liquid to gas, liquid to gas, or liquid to liquid. In all of these cases, the fluid thermal conductivities, k_t, W/m-K, are fairly large on at least one side (generally over 0.2 W/m-K), or phase change is present to drive small-scale turbulence on one side. Gas-to-gas recuperation is sometimes seen in air preheaters in steam power plants. Here, ε is generally below 70%, partly because the minimum flue-gas exhaust temperature is usually limited to ~400 K to reduce corrosion from acid condensation.

Achieving high ε in gas-to-gas exchange with low pumping power has been challenging because volumetric specific heats are much lower than seen in liquids, and gas thermal conductivities are usually low.

Representative examples of compact recuperators include the stainless counter-flow shell-and-tube products by Exergy, as shown in **Figure 1**, with tube diameters of 2.5 to 3.3 mm and tube lengths of 120 to 440 mm [*3*]. Still, to achieve over 96% effectiveness between typical gas flows generally requires connecting about five such modules in series, with resulting high pressure drops and high cost.

Figure 1. A typical, compact, shell-and-tube recuperator by Exergy.

The brazed plate-fin recuperator, such as shown in **Figure 2**, sees more utilization (for example, in recuperated Brayton cycles), but here effectiveness is seldom higher than 84% – largely because of limitations arising from axial conduction, flow maldistribution, and thermal stresses [*4*]. Minimum passage dimensions are typically about 2 mm, and differential pressure handling at temperatures above 480 K is usually quite limited. Again, a series/parallel arrangement could permit high ε and acceptable pressure drop, but again at high cost. The mass would likely be somewhat greater than that for the Exergy compact shell-and tube example but the cost somewhat less. However, plate-type exchangers are not nearly as tolerant of the cyclic thermal and pressure stresses that would be seen in the application we are particularly interested in – synthesis of fuels from renewable (highly variable) energy and waste CO_2 [*5*].

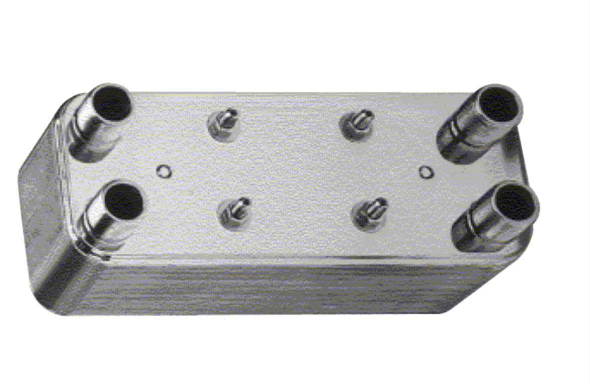

Figure 2. A brazed-plate recuperator from FlatPlate.

An alternative to the above that has seen relatively little usage is the rotating honeycomb regenerator, as used in some turbine engines where system mass is critical. Regenerators have seen little usage largely because of the difficulties in obtaining adequate isolation between the high-pressure and low-pressure streams and because of the shedding of ceramic particles, leading to turbine abrasion. Another factor often limiting their performance is carryover. The relevant design theory has recently been reviewed [*6*].

Optimized, compact high-ε gas-to-gas recuperators require low flow velocities (several percent of the sonic velocity), total flow-path exchange lengths in the range of 0.1 to 2 m, and passage hydraulic diameters of 0.5 to 3 mm, with the larger diameters corresponding to pressures near 0.1 MPa and the smaller sizes corresponding to pressures above 0.5 MPa [*4*]. They have also required the use of construction materials having fairly low thermal conductivity to limit conduction from the hot end to the cold end, though that is not required in the design presented here.

2. The GAS-TO-LIQUID (GL) HIGH-ε MODULE

The basis for the recuperator presented here begins by learning from the highly developed liquid-to-gas exchangers best exemplified in air-conditioning (**AC**) condensers/evaporators and automobile radiators. To achieve high ε in gas-to-liquid recuperation, it is first necessary to arrange several of such exchangers in series, with the liquid flowing serially in one direction and gas flowing in the opposite direction as shown in **Figure 3**. These serially connected cores can be placed inside an insulated pressure vessel to form a Gas-to-Liquid (**GL**) module. However, our goal is to achieve essentially complete transfer of ***exergy*** from the gas to a heat transfer liquid (HTL), or *vice versa*, which requires minimizing exergy destruction or irreversibilities along the *entire* flow path. This in turn requires the following:

1. the heat capacity rates (again, W, or GC_P , W/K) of the gas and HTL streams must be similar;
2. there cannot be any significant amount of phase change within either stream;
3. the fluids should have fairly similar temperature dependence of their specific heats over the applicable range.

The above requirements are obvious upon reflection, as violation of any would lead to a non-uniform temperature difference between the two streams, which clearly prevents the temperature difference from approaching zero everywhere.

There are two additional structural requirements:

1. there must be a high degree of flow uniformity (macroscopically) transverse to the flow direction;
2. the adjacent cores must be effectively thermally isolated.

The first of the above is seldom overlooked in the design of high-ε recuperators, and for the GL module it essentially means there must be minimal variation in the fin pitch across the face of each core. The second of the above is of little consequence in low-ε exchangers. Hence, the fins in multi-row cores are often continuous from one row to the next (see, for example, Armstrong Duralite Plate Fin Coils, [*7*]). However, a low-resistance thermal conduction path from the hot end to the cool end is catastrophic in high-ε recuperators [*4, 6*]. Expressed more quantitatively, the thermal conduction between adjacent cores should be small compared to $W(T_H$-$T_C)/n$, where n is the number of cores in series.

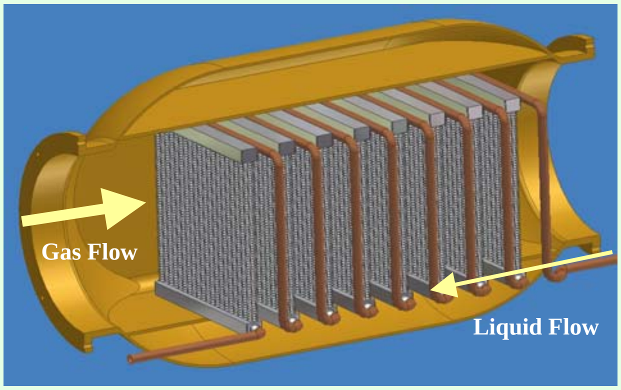

Figure 3. A series of eight cross-flow cores placed inside a pressure vessel to form one counter-flow gas-to-liquid (GL) module.

The heat transfer capacity rate Q_T [W] for a recuperator, when no phase change is present is often expressed by

$$Q_T = U \times A \times T_\delta \qquad (2)$$

where, U is the overall heat transfer coefficient in W/m^{2}K, A is the nominal surface area available for heat transfer and T_δ is the mean temperature difference between the two streams. Increasing the heat transfer rating of a recuperator has generally focused mostly on increasing the effective area A. For a gas to liquid heat exchanger the heat transfer coefficient on the shell side (gas), h_S, is the limiting factor, which motivates efforts to increase the shell side area A_S .

The product of the shell-side heat transfer coefficient h_S and A_S is readily made very large by the use of closely spaced fins, as h_S and A_S will each be approximately inverse with the fin spacing. The minimum fin spacing from a shell-side pumping-power-loss perspective is often approximately inverse with the square of the shell-side pressure under laminar flow conditions, which will dominate in compact heat exchangers [*1*, *4*, *6*]. Hence, from this perspective, the limiting "power compactness" (Q_T/volume) could have a fourth-power dependence on shell-side gas pressure, at least over some range of pressures for clean gases. In practice, for pressures above about 0.3 MPa, the minimum fin spacing will usually be limited by manufacturing tolerances and the need to maintain uniform gas flow over the face of the core. However, fin spacing below 1 mm is practical, which gives shell-side *compactness ratio* of 2000 m^2/m^3 and h_S about 200 W/m^{2}K for air at ~500 K. This represents a power compactness an order of magnitude greater than what is often seen in recuperators.

The flow uniformity challenges are an order of magnitude less demanding than normally seen in high-ε recuperators because of the opportunity for efficient turbulent mixing between each separate core. Moreover, the total exchange-path-flow length is easily kept quite short for the gas (under 2 m, even with 60 cores in series), so the viscous losses can be very low.

The tube-side heat transfer coefficient h_{TL} [W/m^{2}K] is easily made sufficiently large (over 10,000 W/m^2-K is often practical) by judicious selection of the HTL and its flow velocity.

3. The CRLI

A diagram of the compound recuperator with liquid intermediary (CRLI) for gas-to-gas recuperation, using four GL modules in series is shown in **Figure 4**. Here, heat is transferred from a hot gas stream to a cold gas stream of similar heat capacity using two separate HTL loops operating an appropriate temperature ranges for the chosen liquid. Four GL modules, represented by dotted lines, (each with just three cores for drawing simplicity) are shown in this figure. (A more typical actual number of cores per GL module would be 15 to 50.) Essentially all the exergy is transferred from one gas to the HTL, and then from the HTL to the other gas.

A liquid-coupled concept using two banks of series connected GL modules, similar to that shown in Figure 4, has previously been described and analyzed [*8*]. However, such compact heat exchangers would represent a major departure from industrial practice [*9*].

The benefits of this approach compared to shell-and-tube exchangers may not be immediately clear when evaluated primarily on the basis of surface area available for heat transfer. The shell-side area per unit volume increases by a factor of about 5 to 10 but the heat has to be transferred twice. However, the shell-side heat transfer coefficient h_S will also typically increase by a factor of 4 to 10 because the passage thicknesses are decreased. (But viscous losses are still very low, as explained elsewhere.) Hence, total benefit on the shell-side can be a factor of 20 to 100. By selection of an optimum HTL and flow velocity, the tube-side h_{TL} can easily be made over 30 times that of most gases – usually without adding tube-side turbulators. Hence, the CRLI can permit a very large improvement (for comparable powers, flow rates, ε, and pumping losses) in compactness compared to shell-and-tube exchangers for gas-gas exchange even though the heat has to be transferred twice.

Approaching near ideal effectiveness requires a non-linear increase in the number of serially connected cores. To achieve high ε, the exergy destruction must be minimized, which implies that the temperature difference driving the heat transfer at all locations in the exchanger must be as small as practical. For the simplified case of constant C_P over the entire temperature range for both fluids, this effect is shown in **Figure 5**. Calculations show that for practical conditions, ε = 92% may be attained with 22 cores in series, ε = 95% will require 36 cores and ε = 98% will need 93 cores in series.

4. COST EFFECTIVENESS

The strongest benefit of the CRLI is expected to be in the cost, as the cross-flow cores are produced in very high volumes at very low costs. The cost in terms of $/W for the CRLI at high ε can be an order-of-magnitude less than for shell-and-tube exchangers, as we'll show.

A reference point from publicly available data is a good place to begin. The micro-shell-and-tube exchanger model 486-3 by Exergy (containing 127 parallel 3.2-mm tubes, similar to that shown in Figure 1) would achieve 1.7 kW transfer for counterflow air-to-air exchange at 0.9 MPa, 500 K (k=0.04 W/m-K), with T_δ = 25 K at 0.03 kg/s. Its exchange surface area is 0.63 m^2, so its effective U = 110 W/m^2-K (an impressive number for air-to-air recuperation). With 6 modules in series between 700 K and 320 K, one can achieve 93% effectiveness (T_δ = 27 K) with 52 kPa total pressure drop (mostly on the shell side). For moderate quantity purchases, and ignoring the cost and mass of the plumbing connections, this translates into about \$1.8/W and about 2.7 kg/kW.

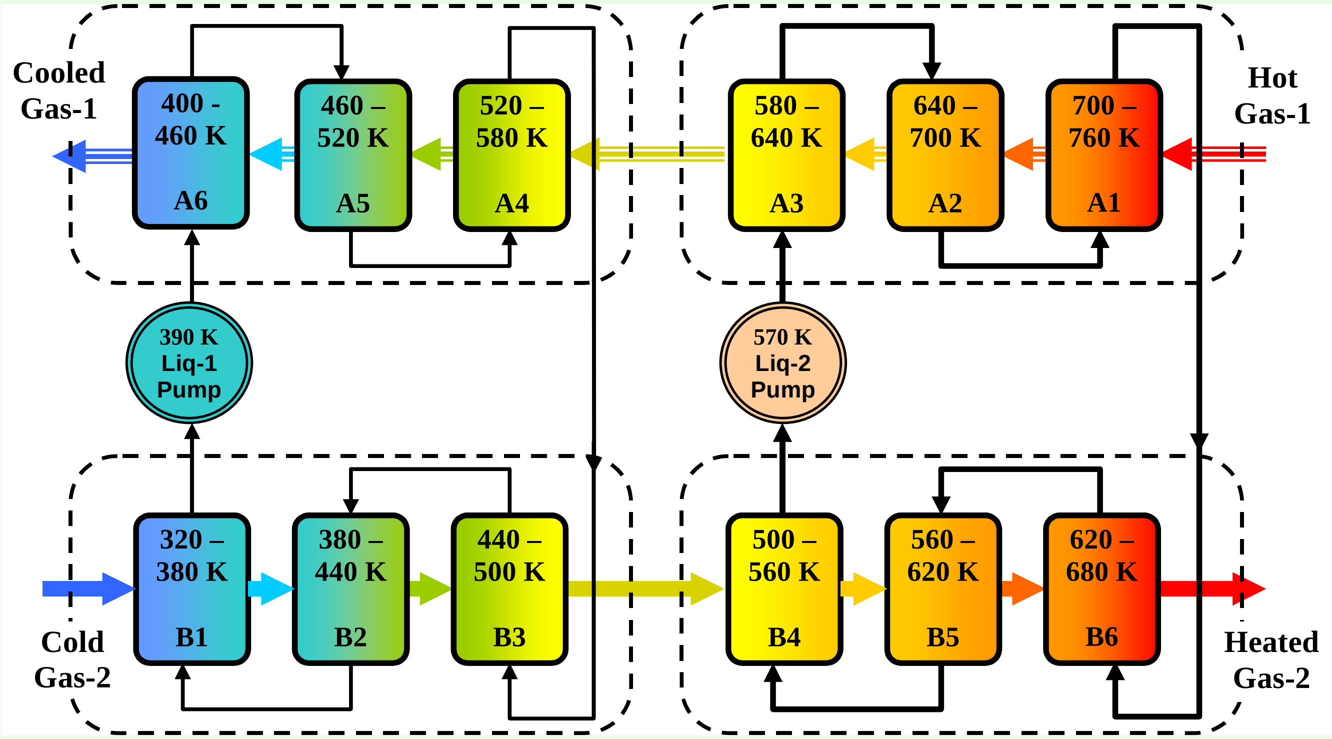

Figure 4. An array of 4 GL modules containing 3 cores each, with two heat transfer liquid loops.

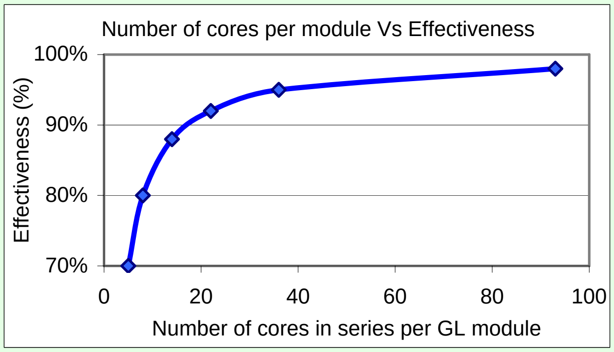

Figure 5. Effect of increasing number of series-connected, thermally isolated cross-flow cores on effectiveness.

For initial estimation purposes, we found that Hayden Automotive oil cooling coils of dimensions about 19 x 30 x 2 cm^3, with 2 mm fin pitch (center-to-center spacing), are locally available at single-piece retail prices of about \$40 each. Using these off-the-shelf parts, the cost of the 48 finned-tube cores needed to make four 12-core GL modules for a CRLI having performance (Q_T and ε) similar to the above example of six series Exergy modules for similar conditions (pressures and flow rates) is about 10% of the cost of the Exergy modules in moderate quantities.

Of course, the above highly simplified comparison has many limitations which are not easily quantified, but useful comments can be made. For the Hayden Automotive cores to be usable above about 450 K, it would be necessary to change the material to copper, and further but straight-forward changes would permit operation to 700 K. Reducing the fin pitch from 2 mm to 1 mm should allow nearly a factor of 4 reduction in the core cross-sectional area required for the same conditions and effectiveness. (Pressure drop may then be comparable to that seen with the tube bundles.) For this temperature range (320-700 K), the CRLI might use two liquid loops, as shown in Figure 4, in which case it would require four pressure vessels, two liquid pumps, and two liquid reservoir/expansion tanks (not shown in Figure 4). However, HTLs are available, as discussed shortly, that would accommodate this full range, and that would allow the CRLI to be implemented with a single pump and 2 GL modules.

Finned-cores more suitable than oil coolers are ubiquitous in air conditioning (AC) condensers. Although these cores are usually intended for operation near 310 K, they are sometimes constructed using copper tubing with aluminum or copper fins brazed on using a filler material having liquidus near 870 K. It is also not too uncommon to see 90Cu-10Ni alloy C706 for the tubing with copper fins. In larger sizes these cores typically use tubing of 9 to 13 mm diameter, and the fin pitch is often about 1.2 mm. Fin length in the direction of air flow is typically ~25 mm per row.

Single-piece prices were obtained from an AC manufacturer for several common, large, AC condensers utilizing four or more finned-tube cores. For example, the retail price for a condenser intended to reject approximately 80 kW with about a 10 K mean temperature differential was about $4000, from which we estimated the cost of the cores alone (which each used two rows of finned tubes) would be about $0.04/W [10]. For heat transfer between air at 500 K (where k is greater than at 300 K by a factor of 1.5) and a liquid with $T_\delta = 13.5$ K, a reasonable estimate of the core-cost would also be $0.04/W, as the copper-alloy materials will be more expensive. The core mass (C706 alloy tubing, C15720 for the fins in the highest-temperature cores, see section **5**) would be about 4 kg/kW.

For air-to-air transfer using a CRLI, the heat in the liquid then must be transferred to the second air stream at similar core cost, and the balance of the system (pressure vessels, pumps, tanks, fluids, plumbing..) must be added. Hence, a reasonable estimate of the cost of CRLI air-to-air exchange for $T_\delta = 27$ K, or 93% effectiveness between 320 K and 700 K is about $0.15/W at sufficient size (perhaps above 500 kW); and the total mass would be about 12 kg/kW. As noted earlier in section 2, the fin pitch should ultimately be able to be reduced by a factor of two (to 0.6 mm) for clean reducing gases above 0.5 MPa with acceptable pressure drops [11], and this may permit a factor of three reduction in mass per power – as well as a reduction in cost per power, as materials costs dominate.

The example using stainless micro-tube modules for similar effectiveness and conditions estimated $1.8/W using moderate-quantity prices – but ignoring the cost of the plumbing for the series connections between modules. In large quantities and including the plumbing costs, it appears the cost by this approach would be about $1.5/W and the mass would be about 3.5 kg/kW. Hence, though the mass of the CRLI using current technology may be 3.5 times greater than with micro-tube modules, it appears the CRLI would cost an ***order of magnitude less*** for the same effectiveness in large applications.

5. HTL SELECTION

The primary requirements for the HTL are chemical stability at the relevant conditions, low viscosity, low vapor pressure, high thermal conductivity, fairly low cost, low health hazard, and high autoignition temperature (**AIT**), though inert gas pressurization would usually be used. It is also beneficial to have freezing point above the minimum start-up temperature, though thaw-out measures can be implemented.

The (turbulent-flow) tube-side heat transfer coefficient may be calculated by [1],

$$h_{\mathrm{TL}} = B_1\, G^{0.8}\, k_t^{0.6}\, (C_{\mathrm{P}}/\mu)^{0.4}\, d^{-1.8} \tag{3}$$

where μ is the dynamic viscosity (cP, centipoise, which is identical to 1 mPa-s, or 0.001 kg/m-s), G is the mass flow rate, d is the tube inside diameter, and B_1 is a dimensioned factor that is nearly constant over a wide range of conditions but varies with surface features and other exchanger design details. (Note: a fluid with $\mu = 1$ cP and $\rho = 1000$ kg/m^3 has kinematic viscosity, μ/ρ, of 1 cSt, centistokes.) A few simple manipulations are useful to collect the fluid properties into a composite fluid property parameter, F_{H}, as follows:

$$G^{0.8} = B_2\, (\rho\, v)^{0.8}\, d^{1.6} \tag{4}$$

$$h_{\mathrm{TL}} = B\, (\rho\, v)^{0.8}\, k_t^{0.6}\, (C_{\mathrm{P}}/\mu)^{0.4}\, d^{-0.2} \tag{5}$$

$$F_H = \rho^{0.8}\, k_t^{0.6}\, (C_{\mathrm{P}}/\mu)^{0.4} \tag{6}$$

$$h_{\mathrm{TL}} = B\, v^{0.8}\, F_H\, d^{-0.2} \tag{7}$$

where the B's are dimensioned constants, v is the mean flow velocity, and F_{H} is a convenient, composite fluid property.

From eq. 7, it might appear that one simply needs to increase the HTL flow velocity to make h_{TL} very large compared to mean shell-side heat-transfer coefficient h_S (as desired for high overall U), but of course that consumes power – which increases almost as the cube of v.

The pumping power also increases with increasing ρ, μ, and flow length. Considering this, a better HTL figure of merit (composite fluid property) for fluid selection than the above F_H would appear to be the following F_{M} [12],

$$F_M = k_t^{0.6}\, (\rho\, C_{\mathrm{P}})^{0.8}/\mu \tag{8}$$

Table 1 lists some properties at 500 K of HTLs that could be selected for different temperature ranges [13]. The column labeled "Risks" gives a single, overall indication of the three hazards normally considered – health, flammability, and reactivity. Other parameters listed include pour point and normal boiling point (n.b.p.). The difference between using F_{H} or F_{M} as a figure of merit is seen to be quite significant. A surprising prediction is that most salts (such as 60NaNO$_3$-40KNO$_3$) may be less optimum than many organics in achieving very high heat transfer coefficient because their viscosity, though it may seem low, is an order of magnitude higher than that of choice organics [12].

One parameter not listed in Table 1 is HTL cost. The most expensive, by far (not including the one not produced for commercial markets, 1-dodecyl-naphthalene) is polyphenyl ether, but it still could be justified in some high-temperature, wide-range applications. However, for operation above 750 K, salts or alloys (see the last three lines in Table 1) would usually be preferred for the HTL. (Tri-o-cresyl-phosphate is the second most expensive HTL in Table 1 produced in commercial quantities.)

Only minor modifications of available AC cores may be needed to permit operation to about 700 K at limited pressures with non-oxidizing clean gases. For reliable operation at higher temperatures, thermal expansion in the tube and fin materials will need to be better matched than in cores designed for operation near ambient temperatures. Operation to 950 K in many non-oxidizing conditions may be possible by changing to an alumina-dispersion-strengthened copper such as C15720 (0.4%Al$_2$O$_3$, bal Cu) for the fins and to the common 70Cu-30Ni alloy C715 for the tubing (~70 MPa yield strength at ~900 K for C715, compared to ~750 K for alloy C706, 90Cu-10Ni).

6. ANTICIPATED APPLICATIONS – *WINDFUELS*

The absence of satisfactory energy-storage solutions has recently led to the increasing availability of negative-priced off-peak grid energy in areas of substantial build-up of wind energy [*5*]. Putting the excess off-peak wind energy into liquid fuels synthesized from renewable H_2 and waste CO_2 should permit complete grid stabilization and at the same time address the need for carbon-neutral fuels for transportation. A process has recently been reported that has strong promise because it is based largely on the commercially proven technologies of wind energy, water electrolysis, and Fischer Tropsch Synthesis (**FTS**) chemistry. However, it also requires an efficient implementation of the so-called reverse water gas shift (**RWGS**) reaction to first reduce the CO_2 to carbon monoxide (CO) and water [*14*].

$$CO_2 + H_2 \rightleftarrows CO + H_2O, \quad \Delta H_{600K} = 38.9\ \text{kJ/mol} \tag{9}$$

Because the above is endothermic, its equilibrium constant is quite low (for example, about 0.25 at 800 K). Therefore, a multi-stage process is required that can efficiently remove the water to very low levels after partial reaction so the reaction can continue. This requires an enormous amount of high-ε recuperation to efficiently cycle the reactants between the reaction temperature and the condensation temperature. However, with the recuperation advance of the CRLI, it should eventually be practical to synthesize fuels, dubbed *WindFuels*, from off-peak wind energy and CO_2 at up to 60% efficiency and at prices that are more competitive than most biofuels [*15*].

Table 1. Some fluid properties of HTLs at 500 K, including normal boiling point (n.b.p.), autoignition temperature (AIT), and two composited figures of merit, F_H and F_M, as defined in the text.

Name	pour point, K	n.b.p. K	AIT, K	ρ, kg/m^3	C_P, kJ/kgK	μ, cP	k_t, W/m-K	*Risks*	F_H	F_M
water	274	373	-	835	4.57	0.11	0.646	0	743	5130
butanol	210	380	699	581	3.78	0.094	0.10	1	179	1260
toluene	190	384	808	640	2.51	0.12	0.077	2	127	657
cumene	130	425	697	661	4.57	0.15	0.108	2	188	1090
ethylene glycol	260	470	673	935	3.16	0.34	0.2	1	221	669
Delo 400 5W40	230	580	620	670	2.5	0.3	0.09	0	100	300
Delo 6170 40wt	255	620	640	680	2.5	0.35	0.09	0	96	260
Mobil 254	212	640	672	700	2.5	0.4	0.1	1	99	250
dioctyl phthalate	250	657	780	798	2.1	0.5	0.11	1	98	200
1-dodecyl-naphthalene	305	676	800	795	2.5	0.41	0.092	1	103	250
tri-o-cresyl phosphate	260	693	680	950	2.2	0.4	0.11	2	136	340
TBPP-100 phosphate	270	708	795	900	2.2	0.5	0.13	1	123	255
polyphenyl ether 5P4E	280	749	860	970	1.9	0.6	0.13	1	114	200
$60NaNO_3$-$40KNO_3$	480	870	870	1950	1.4	4.5	0.45	2	166	77
55Bi-45Pb	400	1800	-	10000	0.15	2.7	4	2	1150	300
38Pb-37Bi-25Sn	400	1900	-	9000	0.18	2.5	8	1	1770	520

The compound recuperator has unique benefits in complex chemical synthesis plants where it is necessary to transfer heat between gas streams that are located far apart, as thermal energy is more readily transferred in a liquid than in a gas. Moreover, with multiple HTL circuits at different temperatures, plant thermal integration can be more readily achieved. There may be some additional benefits when there are highly variable gas flow conditions, such as in a wind-energy-driven RWGS process. Obviously, the HTL flow rate must change to approximately match the varying heat capacity rates in the gases. In cases where the heat capacity rates of the two gas streams may not match for short periods of time, achieving overall high ε with minimal exergy destruction may still be possible with moderate thermal storage on the hot and cold sides of the HTL circuit.

7. CONCLUSIONS

While very closely spaced fins cannot be used with dirty streams or with low-pressure gases, and materials could become expensive for strongly oxidizing gases at high temperatures, there appear to be enormous applications in the offing for high-ε exchange between clean reducing gases at pressures above 0.3 MPa and temperatures below 950 K. For many of such applications above ~100 kW, the Compound

Recuperator with Liquid Intermediary is expected to have order-of-magnitude advantages in cost-effectiveness (W/K/$) compared to conventional recuperators.

SYMBOLS AND ABBREVIATIONS

A – m^2, heat-transfer area
A_S – m^2, shell-side heat-transfer area
B_n – a dimensionless, unspecified constant
d – m, tube inside diameter, ID
ε – thermal effectiveness
C_P – J/kg-K, specific heat
F_H – a fluid figure of merit, see eq. (6)
F_M – an improved fluid figure of merit, see eq. (8)
G – kg/s, mass flow rate
h_S – W/m^{2}K, shell-side heat-transfer coefficient
h_{TL} – W/m^{2}K, tube-side-liquid heat-transfer coefficient
k_t – W/m-K, fluid thermal conductivity
μ – mPa-s, dynamic viscosity, cP, centipoise
Q_T – W, heat transfer rate
ρ – kg/m^3, fluid density
T_δ – K, mean difference between streams
T_C – K, cold gas input
T_H – K, hot gas input
U – W/m^{2}K, overall heat transfer coefficient
v – m/s, mean flow velocity
W – W/K, heat capacity rate

AC – Air Conditioning
AIT – Auto-ignition temperature
CRLI – Compound Recuperator with Liquid Intermediary
GL – Gas-to-Liquid (module)
FTS – Fischer Tropsch Synthesis
RWGS – Reverse Water Gas Shift (reaction)
HTL – Heat Transfer Liquid
NTU – number of (heat) transfer units
TBBP – tri-butyl phenol phosphate (ester)

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